PROGRESS REPORT

PR 91565-430-5

For Month of November 1962

DEVELOPMENT OF AUXILIARY ELECTRIC POWER SUPPLY SYSTEM

NASA Contract NAS 3-2550

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INTRODUCTION

This report covers the work accomplished by Vickers Incorporated under NASA Contract NAS 3-2550 during the month of November, 1962. The objectives of this program are to conduct an engineering study culminating in the design of an electrical power generation system operating on hydrogen and oxygen in a space environment, and to conduct preliminary testing on critical system components.

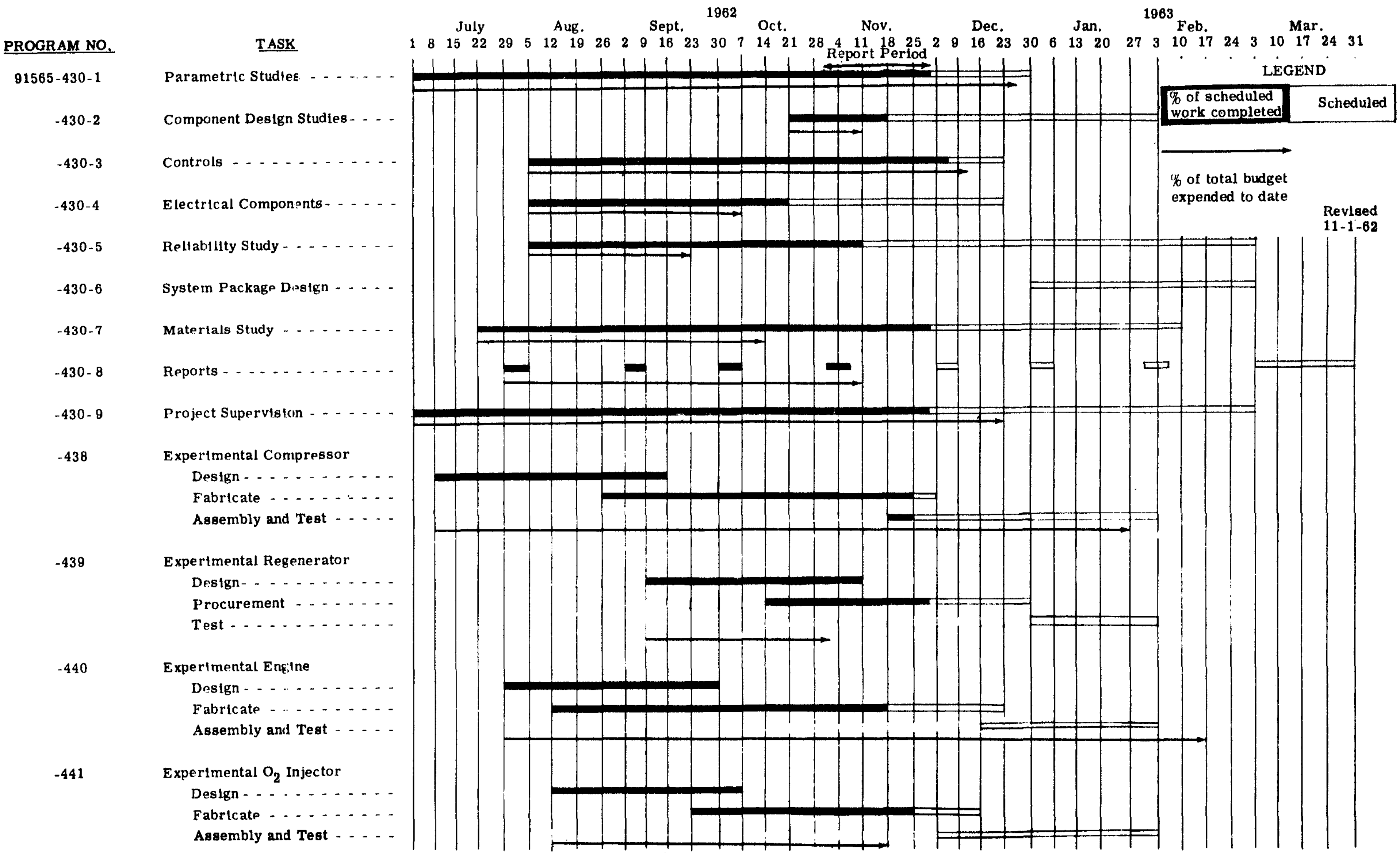
PROGRAM SCHEDULE

The program schedule is shown in Fig. 1. The program plan for this project covering the entries of Fig. 1 was described in the progress report for July, 1962.

The schedule shown in Fig. 1 was revised December 1 in order to reflect delays which have been experienced in the fabrication of experimental hardware. Parts for this program are being given top priority in the prototype shop in order to meet this revised schedule.

Fig. 1

PROGRAM SCHEDULE NASA CONTRACT NAS 3-2550



PARAMETRIC STUDIES

During the past month analysis has been progressing in the areas of cylinder heat rejection, heat sink capacity of the propellants, and the effects of varying O_2/H_2 ratio. A Lunar Excursion Module adaptability study was also undertaken.

Heat Rejection Study

Figure 2 is a graph of engine heat rejection (including 10% of the shaft power output for electrical losses) and heat sink capacity of the hydrogen for various O_2/H_2 ratios.

The following analytical methods were used to calculate the heat sink capacity of the hydrogen and the engine cylinder heat rejection.

The equation for heat sink capacity of the hydrogen expressed in per cent of shaft power is as follows:

$$\% \, S.P. = \frac{W_{H_2} \quad C_{P_{H_2}} \quad \Delta T}{3413}$$

Where W_{H_2} is the flow rate of hydrogen in lbs/hr, Cp is specific heat, ΔT is the change in hydrogen temperature from inlet to 860° R, (the assumed maximum cylinder wall temperature), and the factor 3413 converts BTU/hr to kw.

The engine cylinder heat rejection was calculated by the following equations:

$$q \sim h \quad \Delta T$$

$$h \sim K \quad \left(\frac{MW}{Tm \, \mu} \right)^{.75} \qquad \text{(Ref. 1)}$$

Where h is the film coefficient, K is thermal conductivity, μ is viscosity, MW is molecular weight and Δ T is the temperature differential between mean cycle temperature (Tm) and cylinder wall temperature. Previous analysis (Ref. PR 91565-430-3), on engine heat rejection estimates the cylinder heat rejection to be equivalent to 75% of the shaft power output at an O_2/H_2 ratio of 2/1. Heat rejection was calculated for various O_2/H_2 ratios and then compared to the value calculated for $O_2/H_2 = 2/1$. The total power system heat rejection is the engine heat rejection plus 10% of the shaft power output (to approximate electrical losses). The heat sink available from the hydrogen is shown for two different temperatures, $T_o = 45°$ R and $T_o = 150°$ R. These temperatures represent storage temperature at the beginning and the end of the mission in supercritical tanks.

For a mixture ratio of 2/1, the SPU heat rejection is 85% of the shaft output power and the hydrogen at an initial temperature of 150° R can absorb 34% of the shaft power. This leaves heat equivalent to 51% of the shaft power to be radiated into space by a radiator. For this program only a cursory investigation of radiators and cooling systems can be made and a detailed analysis will be required in a future program. An interesting result also shown in Figure 2 is that at $O_2/H_2 = 1.24/1$, all of the heat rejected by the SPU can be absorbed by the hydrogen at 150 R and no radiator would be required.

Effect of Mixture Ratio on BSPC

The effects on performance of operating at various O_2/H_2 ratios are presented. Since the quantity BSPC (RT_c) is a function of valve timing, BMEP, and engine geometry primarily, it can be regarded as a constant independent of fluid properties, if the variation of specific heat ratio γ is not appreciable. Therefore $BSPC = \frac{BSPC(RT_c)}{RT_c}$

Fig. 2

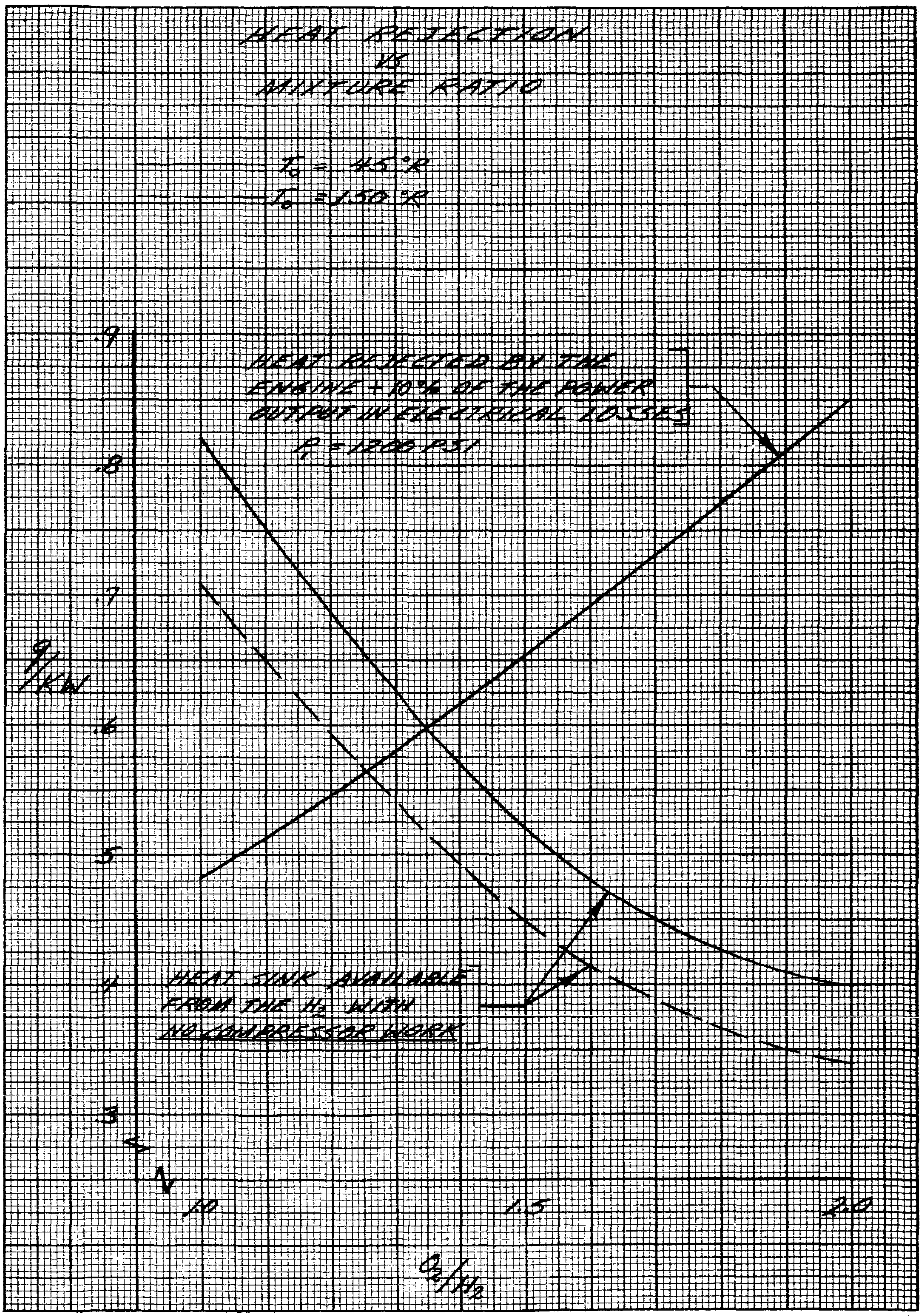
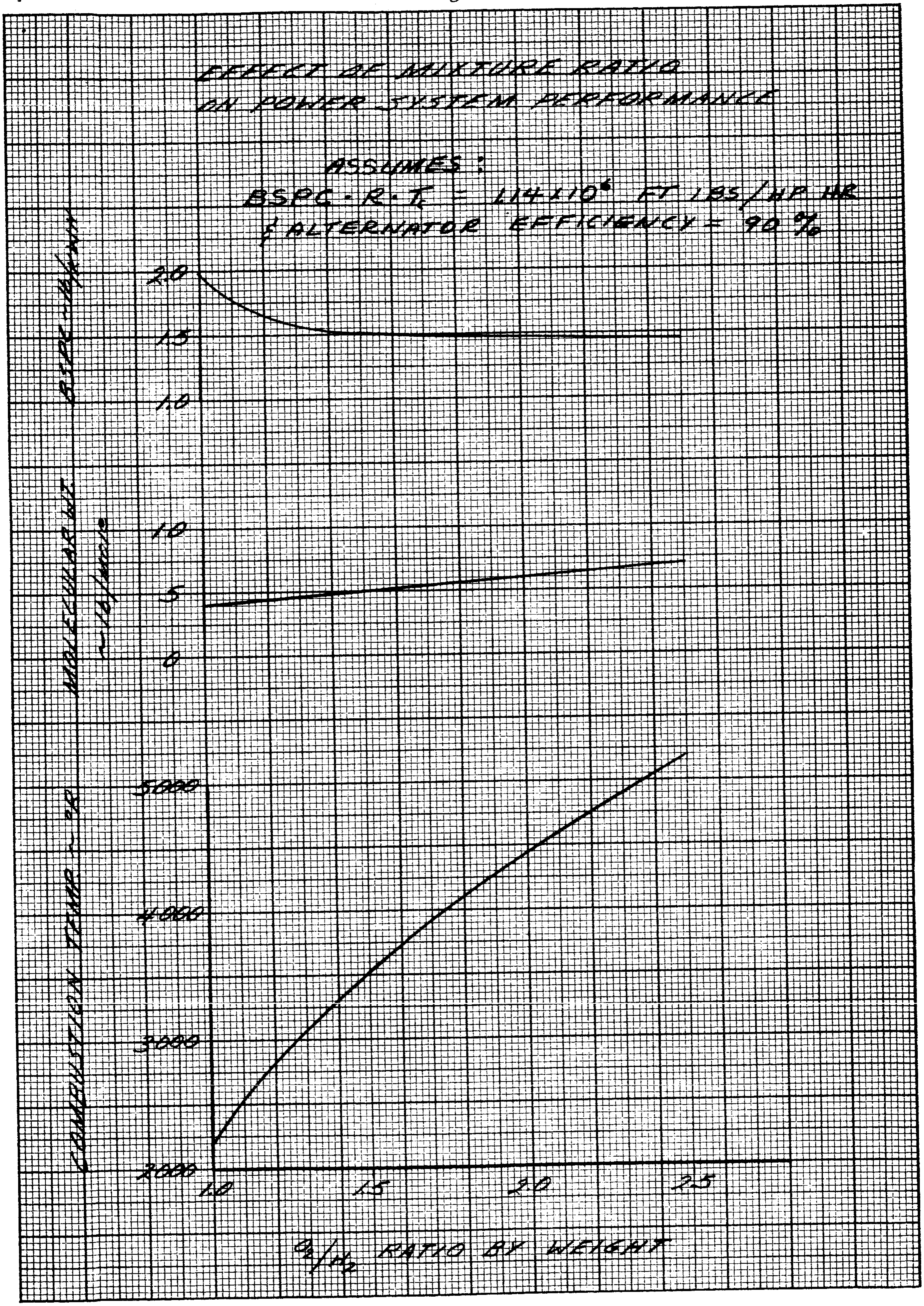

HEAT REJECTION
VS
MIXTURE RATIO
T_o = 45 °R
T_o = 150 °R
HEAT REJECTED BY THE
ENGINE + 10% OF THE POWER
OUTPUT IN ELECTRICAL LOSSES
P = 1200 PSI
HEAT SINK AVAILABLE
FROM THE H_2 WITH
NO COMPRESSOR WORK
q/KW
.9
.8
.7
.6
.5
.4
.3
1.0
1.5
2.0
O_2/H_2

Fig. 3



EFFECT OF MIXTURE RATIO
ON POWER SYSTEM PERFORMANCE
ASSUMES:
BSPC · R · Tc = 1.14x10^6 FT LBS/HP HR
& ALTERNATOR EFFICIENCY = 90%
BSPC ~ lb/kwh
2.0
1.5
1.0
MOLECULAR WT ~ lb/mole
10
5
0
COMBUSTION TEMP ~ °R
5000
4000
3000
2000
1.0
1.5
2.0
2.5
O2/H2 RATIO BY WEIGHT

The variation of molecular weight, T_c and minimum BSPC with mixture ratio for regenerated hydrogen/oxygen cycles are shown in Figure 3. The parameters that are affected the most by changes in O_2/H_2 ratio are operating temperatures, heat rejection, radiator sizes, and propellant tank weight, rather than BSPC.

Mission Adaptability Study

An adaptability study for the Lunar Excursion Module using Grumman's power profile was conducted. The ground rules and assumptions for the engine were the same as those made for the "Earth-to-Lunar Landing Mission". It was assumed that all the propellant was stored at supercritical pressure with the hydrogen at 300 psi and the oxygen at 1000 psi. The Grumman power profile was integrated. The average power for 49.5 hours was found to be 480 watts with 900 watt peaks. To meet this power requirement efficiently the engine displacement must be reduced to .83 cu. in. With a hybrid power modulation system and $O_2/H_2 = 2/1$ the following propellant and tank weights would be necessary. These weights include 10% unusable residual propellants.

	PROPELLANT		TANK		
H_2	56 lbs.		44.8 lbs.		
O_2	28 lbs.		5.6 lbs.		
TOTAL	84 lbs.	+	50.4 lbs.	=	134.4 lbs.

COMPONENT DESIGN STUDIES

The scope of the overall study effort was described in last month's progress report. This effort has been somewhat curtailed due to a reallocation of funds. The effort this month has been concentrated on a flight engine design study and supporting analysis. A sketch of the flight engine will be available next month.

Flight Engine Configuration

The flight engine will operate at a hydrogen inlet pressure of 300 psi and a maximum combustion pressure of 900 psi, using the cycle selected in the parametric studies conducted in this program, and presented in last month's progress report. This low hydrogen inlet pressure permits the use of a single stage hydrogen compressor. It is planned to incorporate this compressor into the crankcase as an integral part of the engine. The compressor piston will be opposed to the power piston of the engine, acting as a dynamic balancer during engine operation. For a discussion of this balancing method see Reference 2.

An oxygen injector similar to the present design NASA Hydrox test engine injector and driven from cams located on the crankshaft will be used. The proposed hydrogen valve will be of similar construction, using a spring for closing rather than a torsional seal tube since hermetic sealing is not necessary. Such a valve mechanism can be lubricated by an oil mist rather than by the pressurized oil supply necessary for conventional valve guides, thereby simplifying oil control.

Power control will be by phasing from 2% hydrogen admission to a 5% admission through a helical spline which moves the cam axially along the crankshaft, changing the orientation relative to the crankthrow.

Throttling below this level by the use of a variable lift mechanism and/or a movable restriction in the flow line is being considered. Variable lift can be accomplished by a variable ratio rocker arm controlled by the same control rod which handles the phase shifting mechanism. All these controls will be mounted compactly along the crankshaft, in a well lubricated region of moderate and controlled temperature.

Flight Engine Analysis

Some preliminary bearing loads and valve flow calculations have been completed and are shown in Figures 4 through 9. The methods of Reference 1 were used to calculate the main bearing loads and the connecting rod force and piston side force at 2% and 5% admission. The results are shown in Figures 4 through 7. A reciprocating mass of 0.5 lb. in an engine of 2.77 in^3 displacement (1.50" bore x 1.57" stroke) were assumed. Gas inlet pressures of 300 psi and peak combustion pressure of 900 psi, without throttling, determined the gas loads. An L/R ratio of 4.0 was assumed.

Inlet hydrogen valve flow is shown in Figure 8. A 30° valve duration following a sinusoidal lift curve, with a valve head diameter of .16 in. and a lift of .040 in. was assumed. It can be seen that this valve size is adequate for full admission. However, it poses a problem for variable - lift throttling, if this is considered.

A poppet exhaust valve is under consideration for the flight engine. Although this valve will add to the overall complexity of the engine, it has several advantages which are enumerated below:

1. More positive expulsion of exhaust gases, with lower compression work. Since recompression is not needed, the valve may be left open until

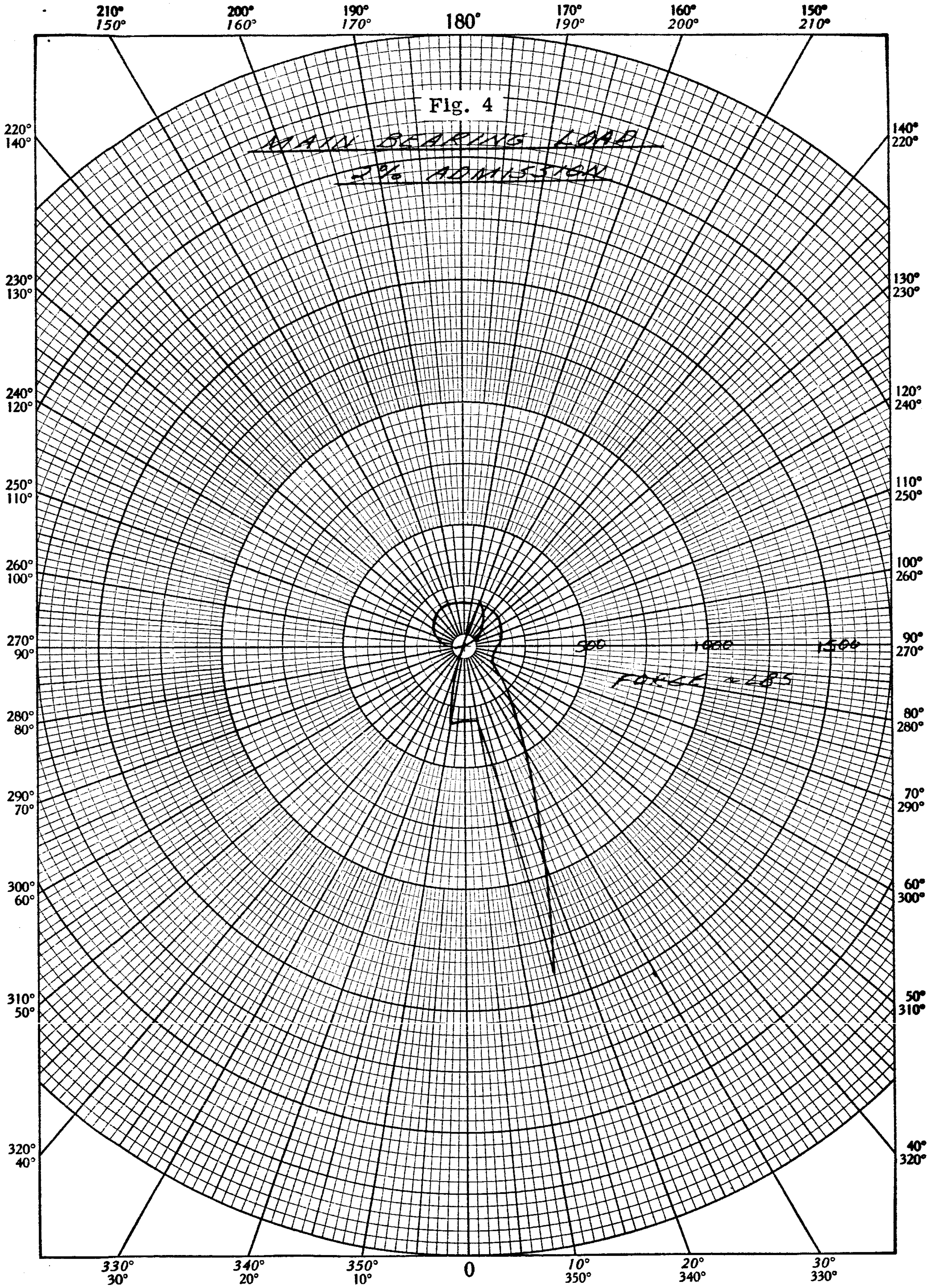


Fig. 4

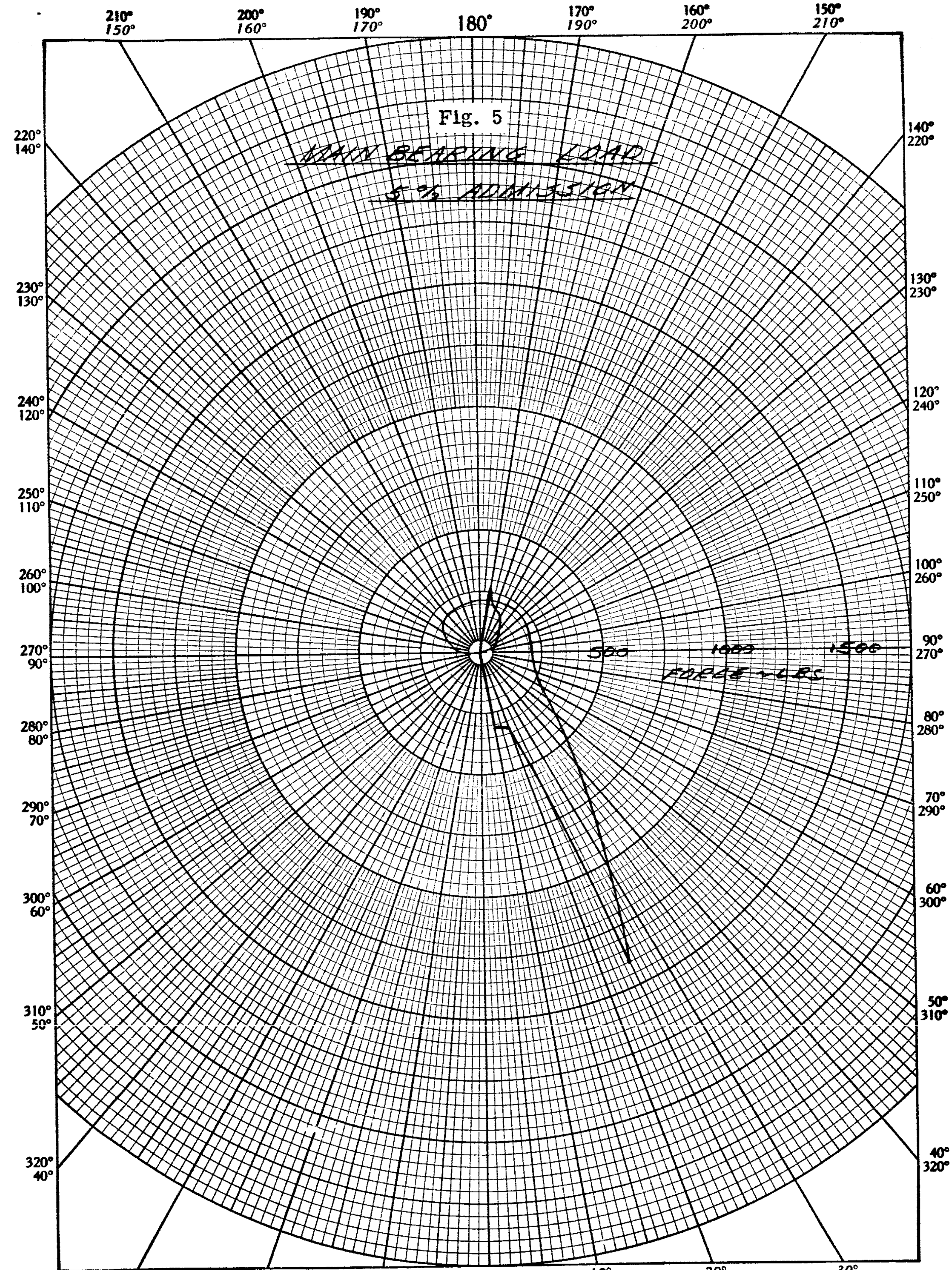


Fig. 5
MAIN BEARING LOAD
5 % ADMISSION
500
1000
1500
FORCE ~ LBS
180°
0
210° 150°
200° 160°
190° 170°
170° 190°
160° 200°
150° 210°
220° 140°
140° 220°
230° 130°
130° 230°
240° 120°
120° 240°
250° 110°
110° 250°
260° 100°
100° 260°
270° 90°
90° 270°
280° 80°
80° 280°
290° 70°
70° 290°
300° 60°
60° 300°
310° 50°
50° 310°
320° 40°
40° 320°
330° 30°
340° 20°
350° 10°
10° 350°
20° 340°
30° 330°

Fig. 6

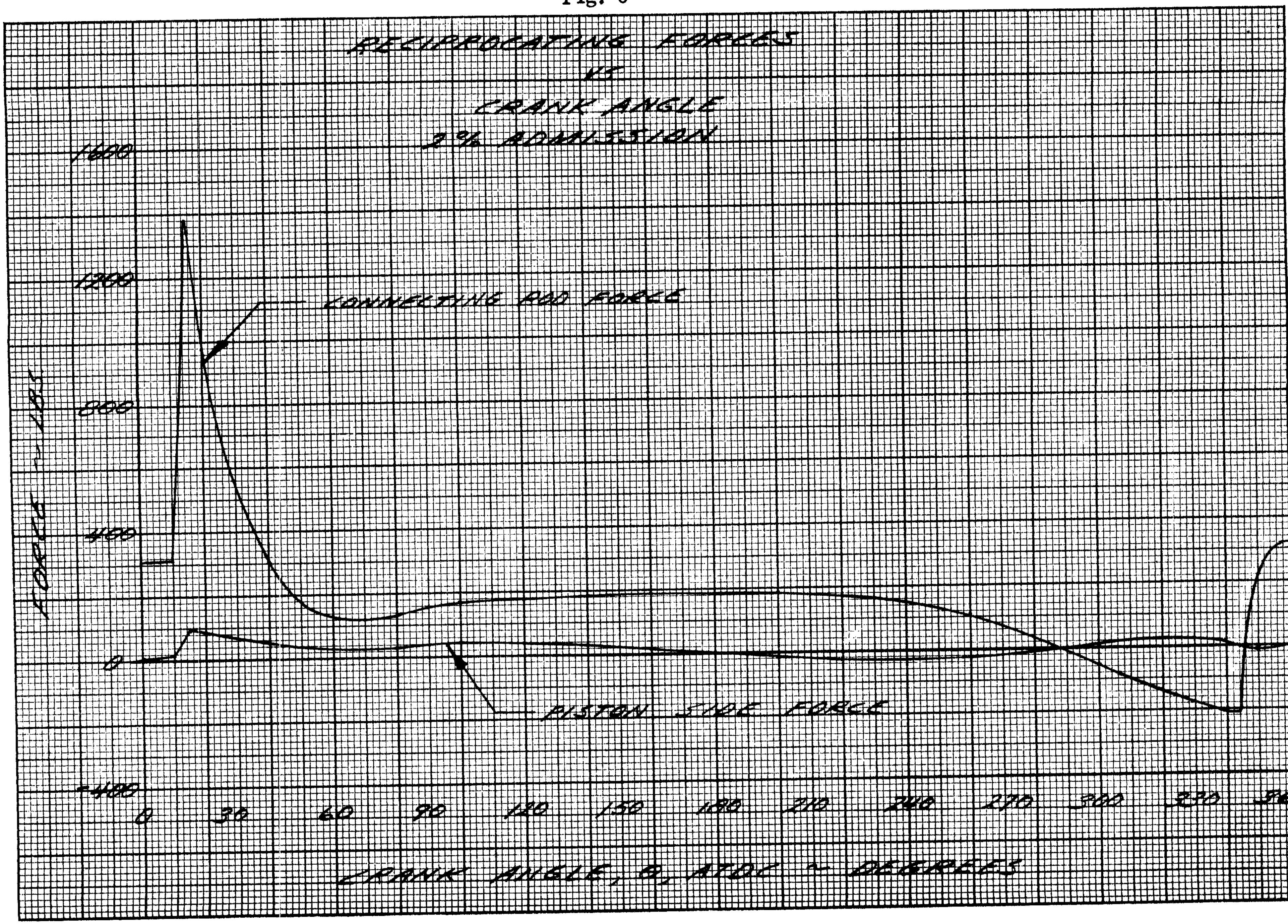
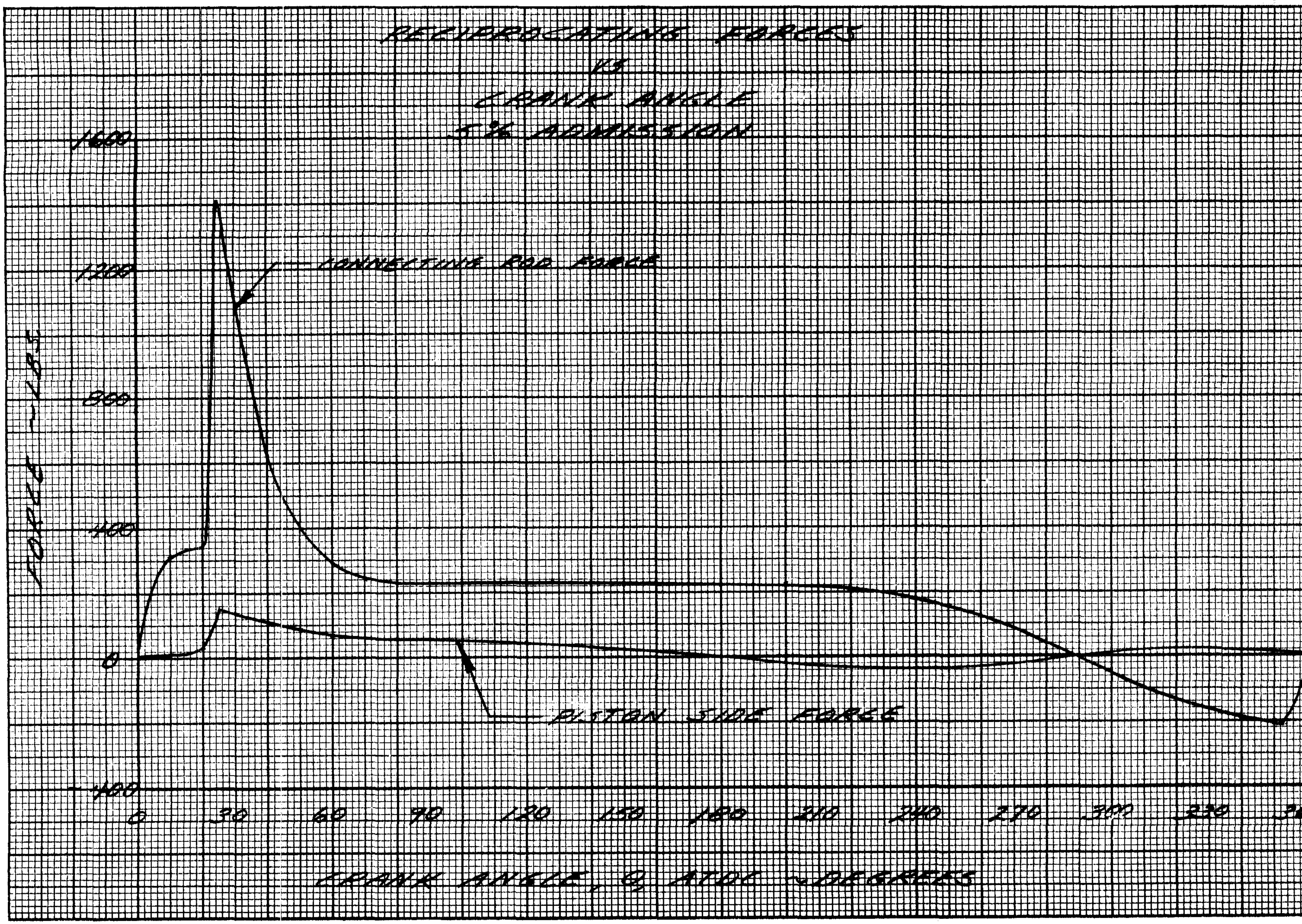
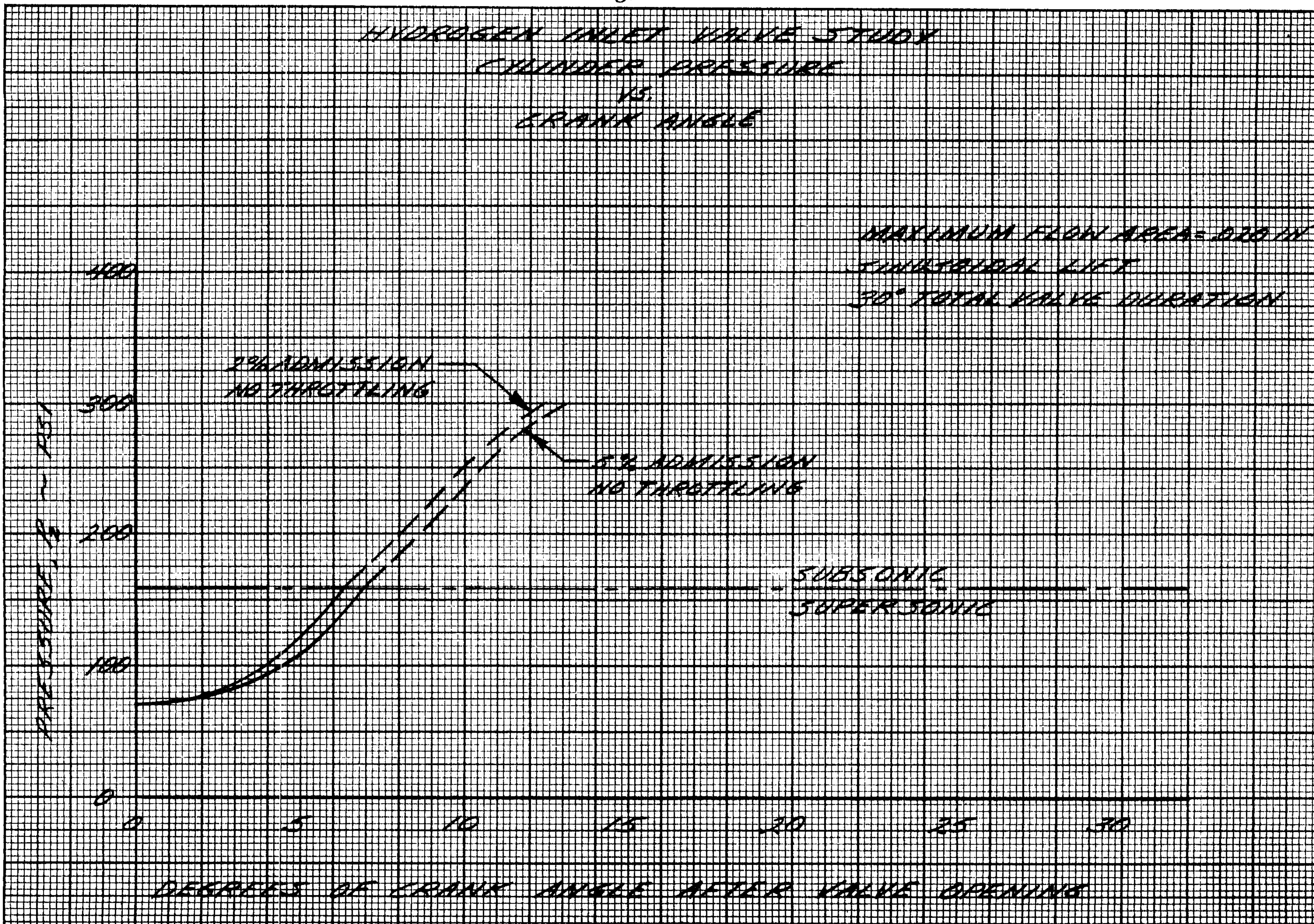

RECIPROCATING FORCES
VS
CRANK ANGLE
2% ADMISSION
CONNECTING ROD FORCE
PISTON SIDE FORCE
FORCE ~ LBS
1600
1200
800
400
0
-400
0
30
60
90
120
150
180
210
240
270
300
330
360
CRANK ANGLE, θ, ATDC ~ DEGREES

Fig. 7



RECIPROCATING FORCES
VS
CRANK ANGLE
5% ADMISSION
CONNECTING ROD FORCE
PISTON SIDE FORCE
FORCE ~ LBS
1600
1200
800
400
0
-400
0
30
60
90
120
150
180
210
240
270
300
330
360
CRANK ANGLE, θ, ATDC ~ DEGREES

Fig. 8



HYDROGEN INLET VALVE STUDY
CYLINDER PRESSURE
VS.
CRANK ANGLE
MAXIMUM FLOW AREA=.020 IN²
SINUSOIDAL LIFT
30° TOTAL VALVE DURATION
2% ADMISSION
NO THROTTLING
5% ADMISSION
NO THROTTLING
SUBSONIC
SUPERSONIC
PRESSURE, P ~ PSI
400
300
200
100
0
0
5
10
15
20
25
30
DEGREES OF CRANK ANGLE AFTER VALVE OPENING

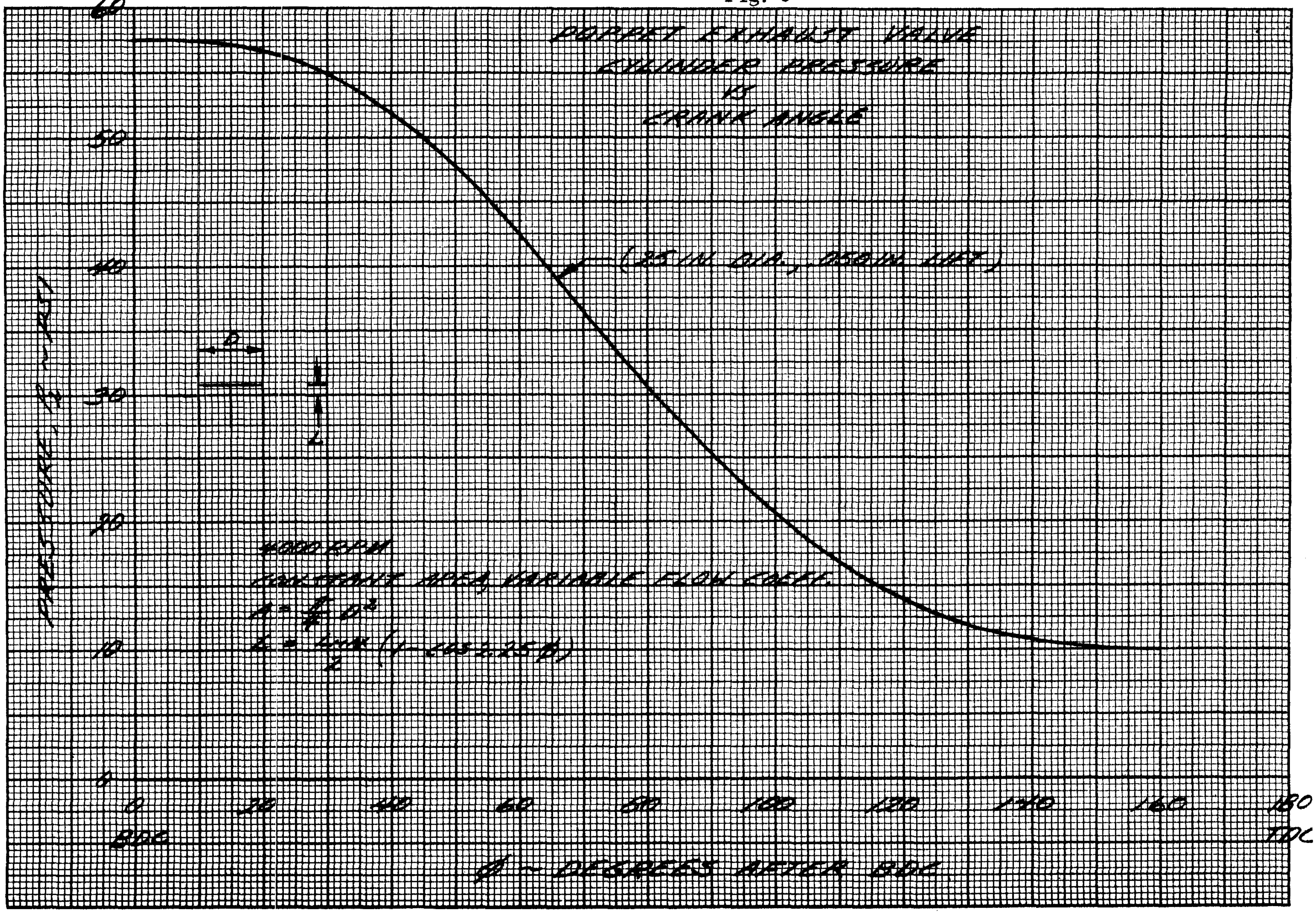
nearly top center. The long duration gives moderate valve dynamic loads.

2. Much better oil control. The circle of ports on the present Hydrox engine may be partially responsible for the loss of considerable oil through the pumping action of the piston rings.

3. More flexibility in the design of the cooling jacket and lower heat rejection to the coolant. Also a more uniform wall temperature is possible, with less distortion of the cylinder bore.

The poppet exhaust valve was sized for minimum flow restriction and a reasonable diameter using the methods in Reference 1. A side valve configuration is considered. Due to the long opening duration valve dynamic loads are very low. Valve flow in terms of cylinder pressure is shown in Figure 9 for the recommended minimum valve size.

Fig. 9


POPPET EXHAUST VALVE
CYLINDER PRESSURE
VS
CRANK ANGLE
(.25 IN. DIA., .058 IN. LIFT)
4000 RPM
CONSTANT AREA, VARIABLE FLOW COEFF.
A = π/4 D²
L = Lmax/2 (1 − cos 2.25 θ)
D
L
PRESSURE, P ~ PSIA
60
50
40
30
20
10
0
0
BDC
20
40
60
80
100
120
140
160
180
TDC
θ ~ DEGREES AFTER BDC

CONTROLS

I. Engine Control Configuration Analysis and Design

Analysis of design requirements is being concentrated on the control configuration shown in Fig. 7 of the October Progress Report, with the introduction of one new design concept. This is the concept of power modulation using variable lift poppet valves in lieu of throttling valves. Much work remains to determine the possibility of the methods proposed. From the standpoint of generating system control requirements, the valve flow requirements may be determined and applied to whichever mechanism proves most feasible.

A brief set of requirements for the system is shown in Table I. The computer block diagram which was used in generating these requirements is included as Figure 11. This block diagram delineates the general form of the system, and specifically relates the various requirements included to the system hardware representation. These requirements are based on meeting transient performance in response to .87 kw step load inputs.

A summary of valve requirements (Table II) is also included. This list does not include the poppet valves which are an integral part of the engine or compressor. Location of these valves can be best accomplished by referring to Fig. 7, October Progress Report.

II. System Performance

Figure 10 is included to show system performance for the speed control loop with nominal values of gain, compensation, etc. The system meets the required response, and a small amplitude limit cycle is exhibited. This is primarily due to valve mechanism freeplay. Unless

Fig. 10 - Hydrox Engine Speed Loop Response, 6000 rpm Generator System

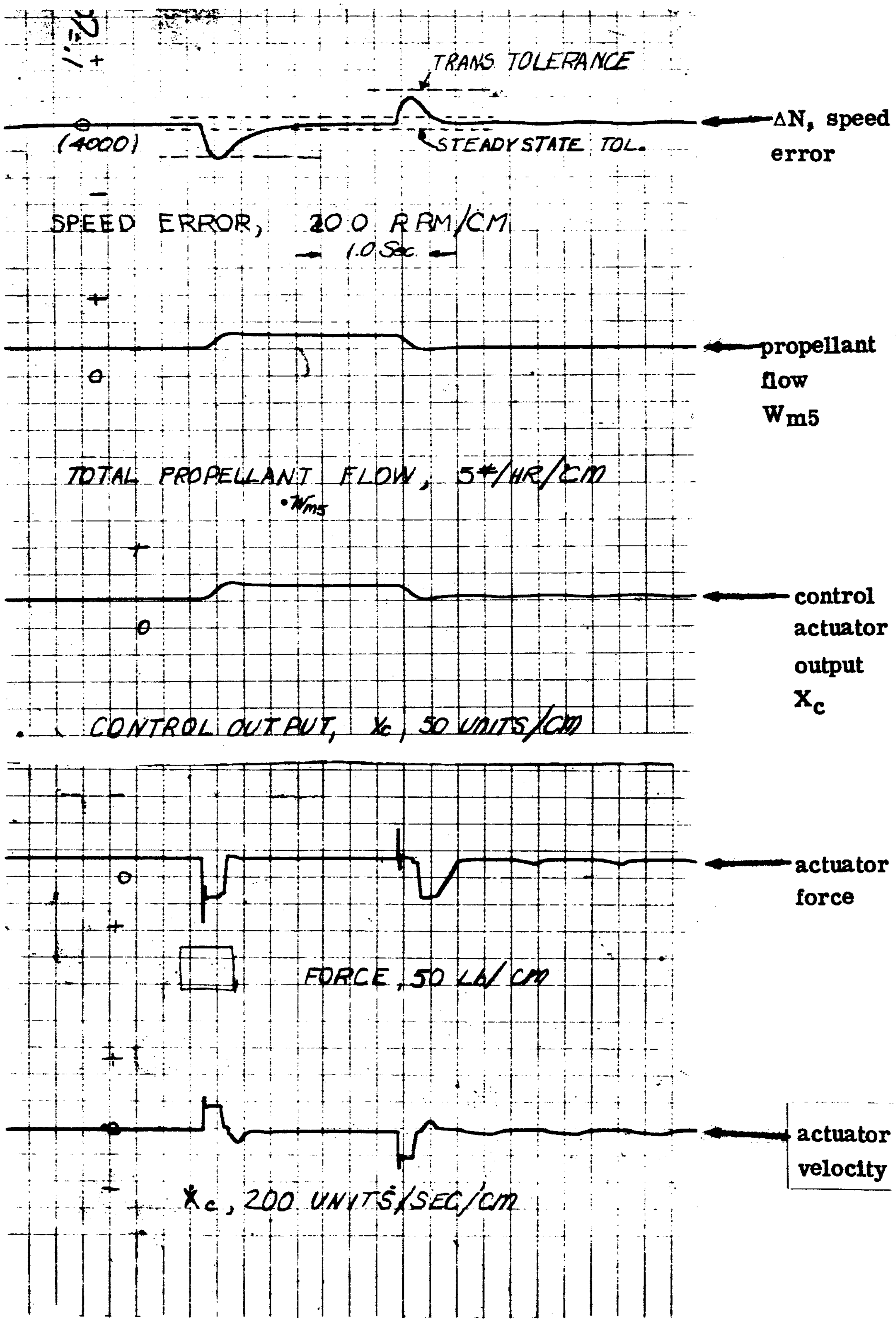
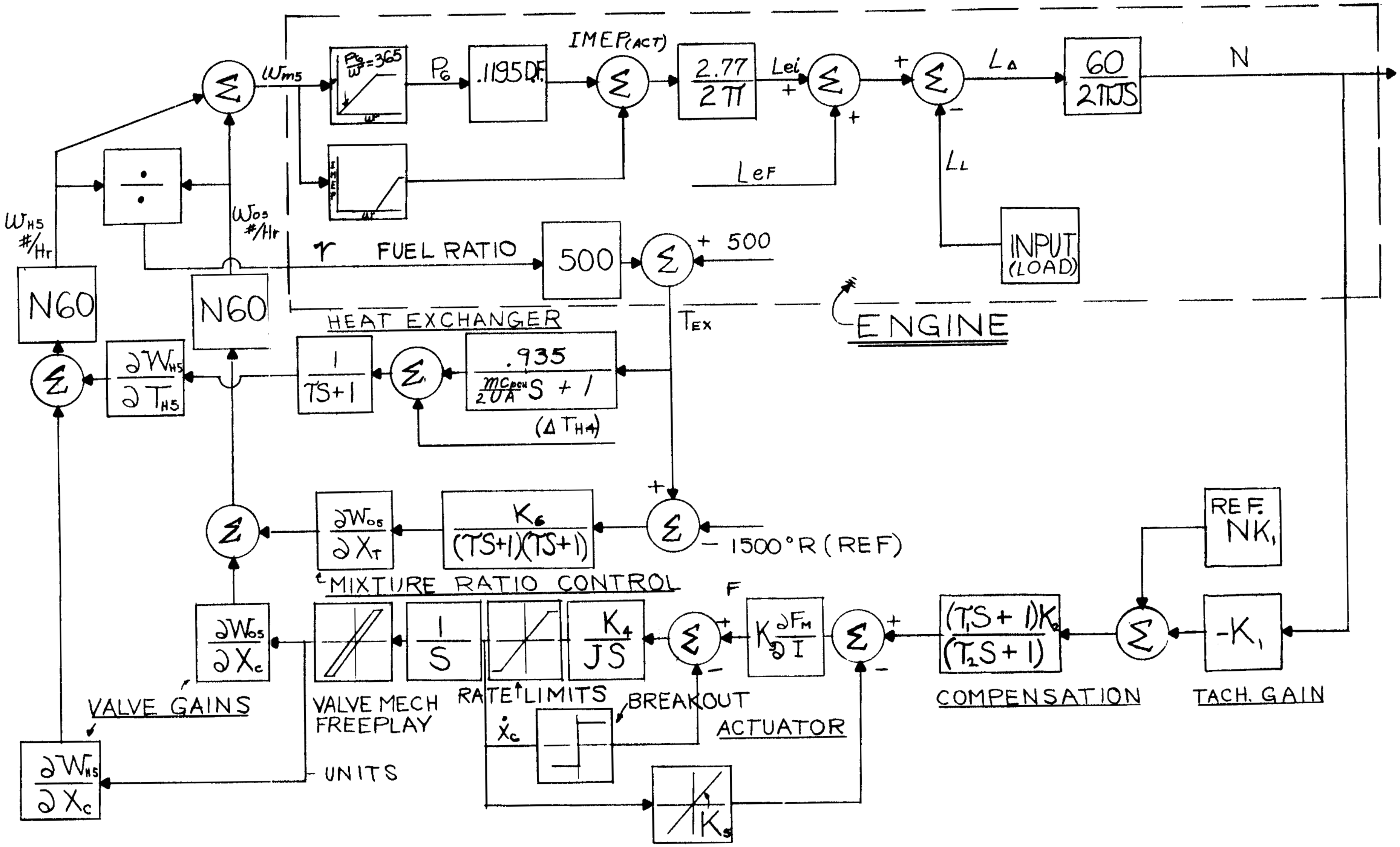


Fig. 11


HYDROX ENGINE COMPUTER
BLOCK DIAGRAM
IMEP(ACT)
.1195DF.
2.77 / 2π
Lei
Lef
60 / 2πJS
N
INPUT (LOAD)
FUEL RATIO
500
+ 500
TEX
ENGINE
HEAT EXCHANGER
1 / TS+1
.935
N60
N60
1500°R (REF)
MIXTURE RATIO CONTROL
REF. NK1
-K1
TACH. GAIN
COMPENSATION
ACTUATOR
BREAKOUT
RATE LIMITS
VALVE MECH FREEPLAY
VALVE GAINS
UNITS
1 / S
K4 / JS

the amplitude of this limit cycle proves undesirable, measures to correct it do not seem necessary. The amplitude is quite likely to be much less than shown, since the accuracy of the valve mechanism will probably be much better than the relatively loose requirements which were assumed.

III. Final Phase of Study

The study will be finalized next month. In finalizing the study, effort will be devoted to obtaining and reducing data on the final system and examining hardware concepts.

TABLE I

Summary of Design Requirements:

Note: System requirements are based upon 44.5 units of controller output to give 100 percent power under most critical set of operating characteristics.

Actuator Requirements (Magnetic Particle Clutch Assumed):

Rate: 6000 rpm Generator	85 Units/Sec. Minimum, Max.
4000 rpm Generator	40 Units/Sec. Minimum, Max.

Force in Excess of Breakout:

10 lbs. (Based upon .017 lb-sec.^{2}/in. mass)

Amplifier and Clutch Gain:

(Based upon .017 lb-sec.^{2}/in.)

Breakout Force	Minimum Gain (K_3 [illegible]) lbs/unit/sec.
± 60 lbs.	20.0
± 50 lbs.	17.5
± 40 lbs.	15.0
± 20 lbs.	7.5
± 10 lbs.	4.0

For Feedback Gain: $1 = K_5$

Valve Positioning System Accuracy:

Maximum Mechanism Freeplay, ± 4.5% smaller valves very desirable. (Or about ± 2 units)

TABLE I - CONTINUED

Compensation Requirements:

$\frac{\tau_1 s + 1}{\tau_2 s + 1}$ gives satisfactory compensation where $\tau_1 \approx .30 > \tau_1 > .167$ sec.
$\tau_2 \approx .0208$ sec. or less

Tachometer and Amplifier Gain ($K_1 \times K_2$):

1 Units/sec/rpm for 6000 rpm system generator

3 Units/sec/rpm for 4000 rpm system generator

Gain variations up to 2 times nominal gain or down to .6 times nominal gain can be tolerated.

Tachometer and Reference Accuracy:

Steady state accuracy, less than .25% for all conditions.

Ripple - must be kept to a minimum under 60 cps.

Mixture Ratio Control Requirements:

Thermostatic actuator gain, K_6 .1 unit °F
.05 units/°F to .5 unit °F - acceptable

Time Constant
Sec, ± .5 sec.

Accuracy,
Sufficient force for mechanism breakout for 15° temperature error.

Valve Requirements - See Chart

REQMT Name and Function	LEAKAGE Internal & External	TYPE	ACTUATION Source of Power	SIZE Estimate Operating Area	PRESSURE OF Supply	TEMP	WEIGHT FLOW Maximum, Minimum	PRESSURE DROP	RESPONSE	MODE OF OPERATION	RESOLUTION OR ACCURACY	COMMENTS
Oxygen Comp. Throttle Throttles O_2 Boiloff Into Comp.	.03 lb/hr External; Internal, Must Operate Under No Flow	Constant Upstream Pressure Regulator	Mech. Regulator Operated By Supply Gas	.0036 in^2 (Min-Max Area) To .000046 in^2	15 to 175 psia Temp 160-200°R	Temp. of Saturated Vapor at Supply Pressure	On 1.15 lb/hr. 2.85 lb/hr. Max. Off Leakage	Output Pressure Constant @ 15 psia	Not Expected To Be Critical Less Than .04 Sec.	Used Under Boiloff Supply Condition	± 1 psi, ± 5 psi For No Flow	
Hydrogen Comp. Throttle Throttles H_2 Boiloff Into Comp.	.03 lb/hr External; Internal, Must Operate Under No Flow	Constant Upstream Pressure Regulator	Mech. Regulator Operated By Supply Gas	.00318 in^2 (Min-Max Area) To .000028 in^2	15 to 175 psia Temp 36-60°R	Temp. of Saturated Vapor at Supply Pressure	On .57 lb/hr To 1.425 lb/hr. Max. Off Leakage	Output Pressure Constant @ 15 psia	Not Expected To Be Critical, Less Than .04 Sec.	Used Under Boiloff Supply Condition	± 1 psi, ± 5 psi For No Flow	
Hydrogen Reservoir Shutoff Valve (May be combined with hydrogen tank valve, 1,2-way valve)	.01 lb/hr Internal .01 lb/hr External	Shutoff Valve	Electrical	.005 in^2 To 0	400 psia ± 10 psi	Approx. 1050°R	1.425 lb/hr. Max.	5 psi Max.	.5 Sec. Max.	Open Under Boiloff, Closed For Tankage	On-Off	
Oxygen Reservoir Shutoff Valve	.01 lb/hr Internal .01 lb/hr External	Shutoff Valve	Electrical	.01 in^2 To 0	1200 psia ± 20 psi	Approx. 1000°R	2.85 lb/hr. Max.	10 psi Max.	.5 Sec. Max.	Open Under Boiloff, Closed For Tankage	On-Off	NOTE: 1. All valves req. to be light weight for space application 2. Downstream volume Not critical on all valves.
Hydrogen Tank Valve Admits Hydrogen To System From Tank	.01 lb/hr Ext. .01 lb/hr Int.	Shutoff Valve	Electrical	.0001 in^2 To 0	400 psia Super Critical Storage	45°R To 140°R	1.425 lb/hr. Max. Off Leakage	5 psi Max.	Not Expected To Be Critical Less Than 1.0 Sec.	Used Under Tankage Supply Condition	On-Off	
Oxygen Tank Valve Admits Oxygen To System From Tank	.01 lb/hr Ext. .01 lb/hr Int.	Constant Upstream Pressure Reg. With Override To Off	Electrical Override To Off Position Mech. Relief	.0001 in^2 To 0	1200 psia Super Critical Storage	190°R To 400°R	2.85 lb/hr. Max. Off Leakage	10 psi Max.	Not Expected To Be Less Than 1.0 Sec.	Used Under Tankage Supply Condition	See Pressure Drop	
Oxygen Reservoir Relief Valve Admits Oxygen To Boiloff Supply Or Overboard	External .05 lb/hr	Constant Back Pressure Regulator	Reservoir Pressure (Mech.)	.0007 in^2 To 0	1200 psia ± 20 psia (Required Pressure)	Approx. 1000°R	1.42 lb/hr. Max.	1000 psi Min.	Not Critical	Boiloff Supply Condition	See Pressure Drop	
Hydrogen Reservoir Relief Valve Admits Hydrogen To Boiloff Supply Or Overboard	External .05 lb/hr	Constant Back Pressure Regulator	Reservoir Pressure (Mech.)	.0003 in^2 To 0	400 psia ±10 psia (Required Pressure)	Approx. 1050°R	.75 lb/hr. Max.	225 psi Min.	Not Critical	Boiloff Supply Condition	See Pressure Drop	

Table II - Hydrox Engine Valves

ELECTRICAL COMPONENTS

Work accomplished in November consisted of investigating cooling methods of the alternator and the associated voltage regulator. One promising cooling method is by operating the rotating elements in an atmosphere of hydrogen at 5 to 7 psi absolute pressure and between -70 and -60° Centigrade.

The temperature of the hydrogen coolant has been selected on the basis of the saturation resistance and forward conduction characteristics of solid state rectifiers used within the coolant atmosphere in the rotating elements of a "brushless" alternator.

A hydrogen gas - incoming high pressure hydrogen heat exchanger incorporated into the structure of the alternator has been proposed as a cooling system. Hydrogen coolant pressure will be maintained against shaft seal leakage by a pressure regulator using bleed off from the hydrogen propellant line.

A final decision on alternator cooling must be based on overall system cooling requirements, and on obtaining the best possible use of incoming propellant heat capacity and temperature.

RELIABILITY

During November, the survey of major engine manufacturers was continued. Letters were sent to the Army Corps. of Engineers, Ft. Belvoir, Virginia, and the Lycoming Engine Corporation, Stratford, Connecticut, requesting failure rate information for small reciprocating engines. No new information has been received from the companies previously contacted.

Because of the limited amount of time allotted to Reliability Engineering in this program, it is expected that most of the remaining effort will be accomplished in the latter part of the program when the system design has been more firmly established.

EXPERIMENTAL OXYGEN/INJECTOR

I. Design - Complete

II. Detailing - Complete

III. Fabrication and Procurement

Fabrication of the Experimental Oxygen/Injector is approximately 75% complete. The current estimated completion date is December 16, 1962.

IV. Facilities

The O_2 Injector test stand is complete except for the addition of a resistance heater. The heat circuit is shown schematically in Figure 12. The complete test circuit is mounted on a portable cart as shown in Figure 13 (photograph).

V. Test on Nitrogen

The currently planned tests on nitrogen are as follows:

A. Leakage - The valve will be checked for external leakage and leakage between the poppet and seat. This test will be performed during assembly.

B. Functional and Mean Flow Tests - The valve will be driven at 67 cycles per second (equivalent to 4,000 rpm engine speed) while inlet pressure is varied from 500 to 1500 psi. Test data will be reduced to mean flow vs. pressure drop.

C. Flow vs. Valve Lift - The valve will be tested to determine flow coefficient variation with lift.

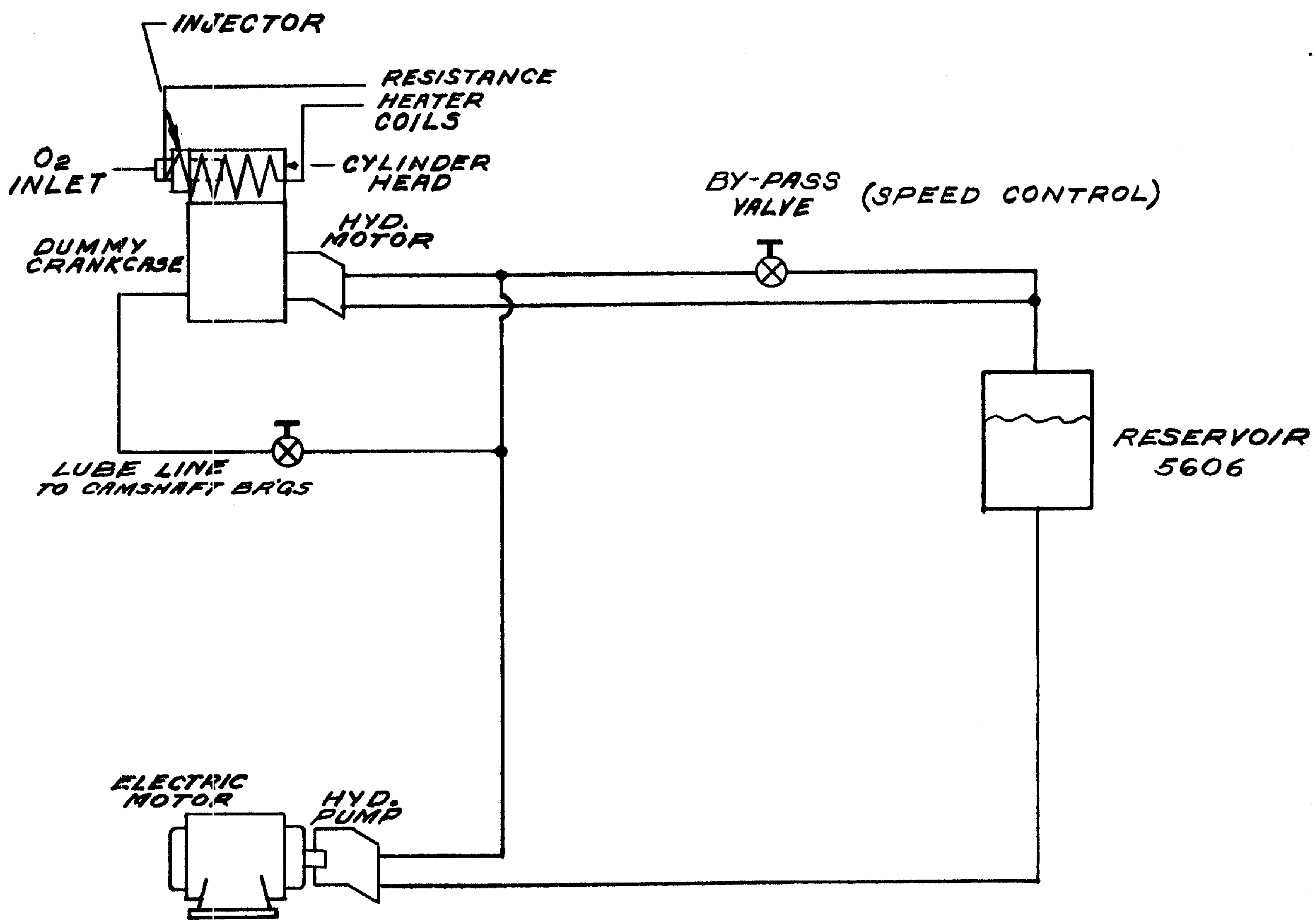


Fig. 12

SCHEMATIC
O_2 INJECTOR BENCH TEST STAND

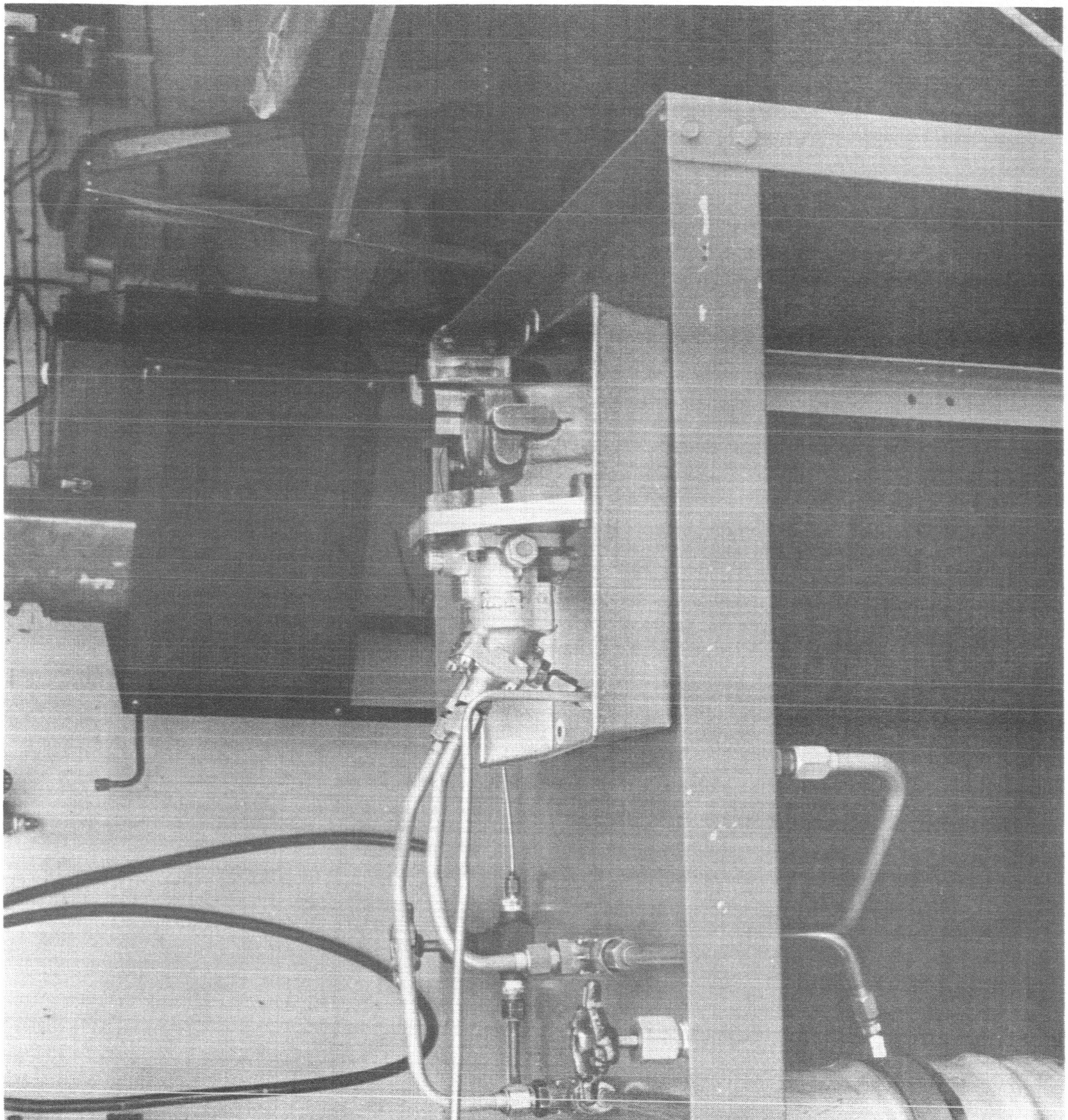


Fig. 13 - Injector Test Stand

VI. Performance Test With Oxygen

The purpose of these tests is to determine the design and material compatibility with oxygen. The currently planned tests on oxygen are as follows:

A. The valve will be run for several hours at 67 cps with ambient temperature oxygen, while the inlet pressure is varied between zero and 1500 psi.

B. Test A will be repeated while the injector body is heated electrically.

During both tests, the injector will be periodically disassembled, inspected, and critical components photographed.

EXPERIMENTAL COMPRESSOR

I. Design - Complete

II. Detailing - Complete

III. Fabrication

Fabrication of the Experimental Compressor is approximately 90% complete. The current estimated completed date is December 2, 1962.

IV. Assembly and Bench Checkout

Assembly of the compressor is in progress. The inlet and outlet valves have been assembled into both the first and second stage head assemblies and the poppet and seats have been lap fitted. The head assemblies are to the right and left in the photograph of compressor components shown in Figure 14. The compressor body is also shown.

V. Test Facilities

The compressor test facilities are approximately 80% complete. The photograph shown in Figure 15 was taken inside the test shed which is next to the control room wall. The stand in the middle of the picture holds the compressor test stand, the hydraulic drive motor, the compressor drive leakage, speed pickup, torque pickup, and flywheel. The nitrogen dewar which will hold the liquid nitrogen to cool the incoming hydrogen is in the lower left hand corner of the photo.

VI. Currently Planned Preliminary Tests on Nitrogen

During these tests both the compressor and test set-up will be checked for: function, mechanical integrity, and unanticipated internal and

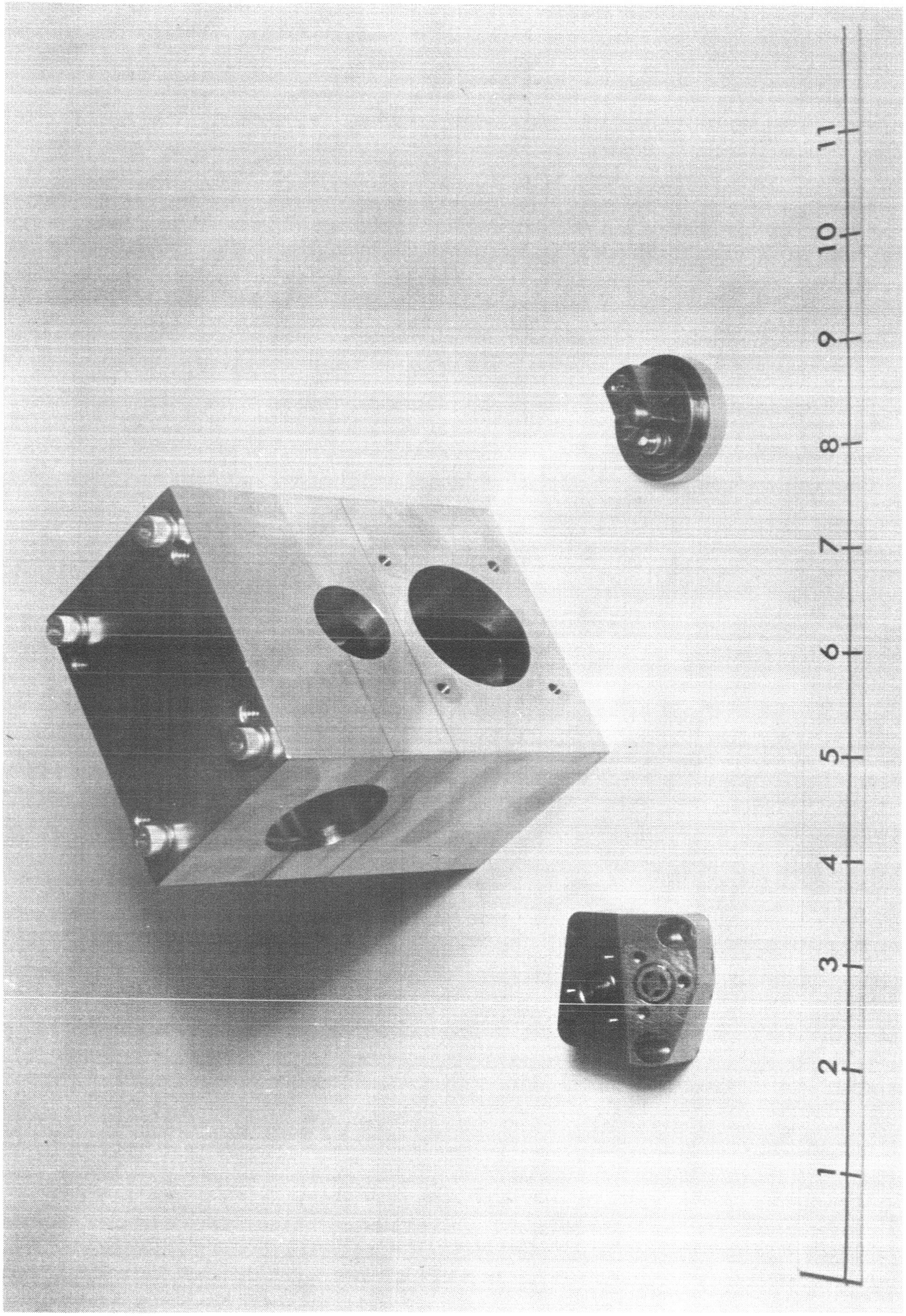


Fig. 14 - Compressor Parts

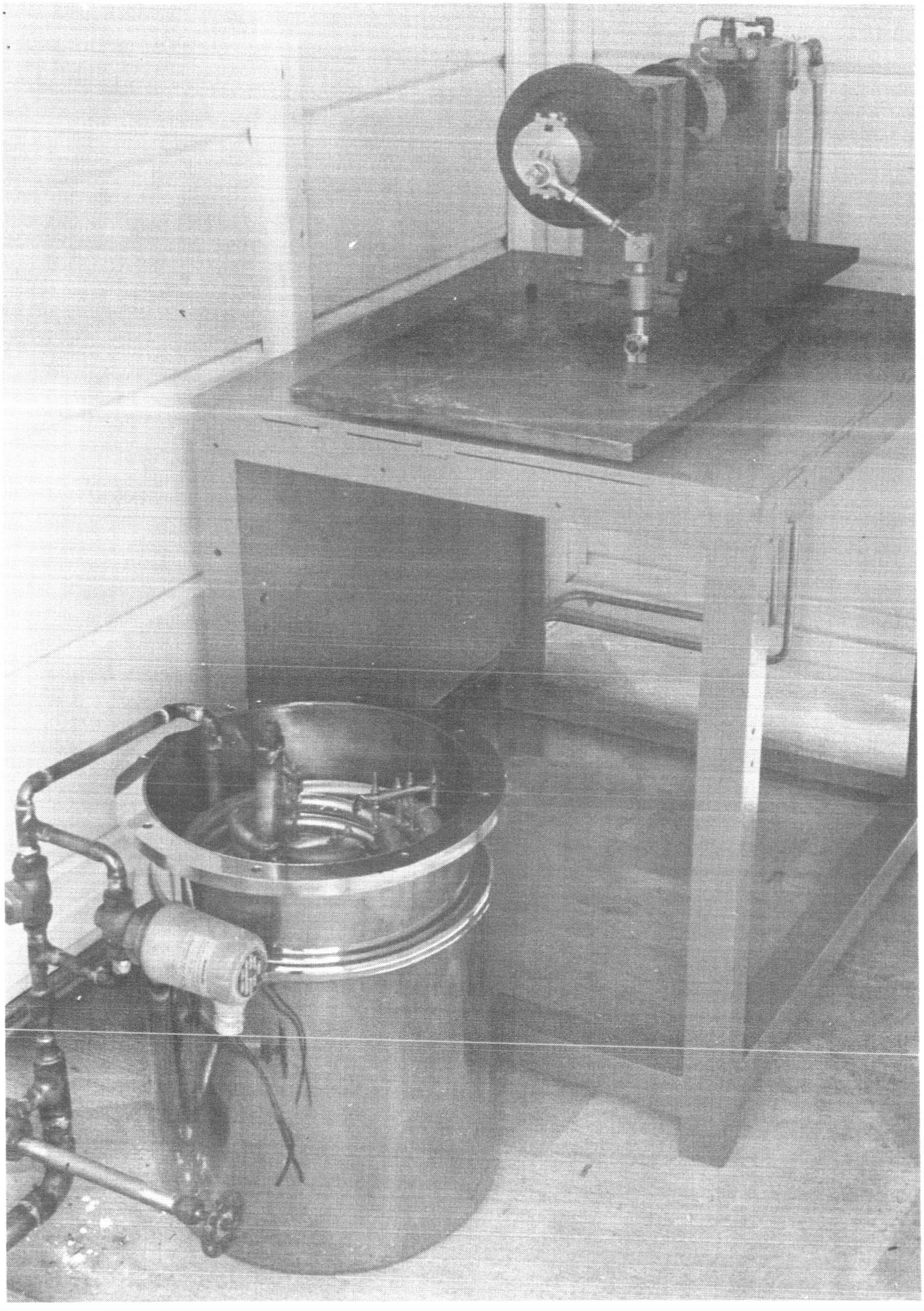


Fig. 15 - Compressor Test Setup

external leakage. The compressor will be run-in, and motoring torque vs. speed data will be taken. If deemed necessary, the compressor will be disassembled, inspected, and adjustment and repairs will be made.

VII. Performance Tests With Hydrogen (Currently Planned)

A. Leakage: Both the compressor and test set-up will be checked for leakage.

B. Design Performance: Test runs will be made with flow rates of 100%, 75%, 50%, and 25% of maximum rated flow (out) where maximum is 1.41 lbs/hr. with, -

Speed = 4000 rpm
Discharge Pressure - 1200 psia
The Inlet Pressure varied to determine flow rate

Sufficient data will be taken to determine efficiency and the power requirements.

C. Flight Design Performance: The second stage will be by-passed and the clearance volume of the first stage will be adjusted (with spacer) to deliver a discharge pressure of 300 psi. Test B will be repeated. Material selection will be verified during testing. Parts will be examined for wear, galling, pitting, etc. Two different material piston rings will be run.

EXPERIMENTAL ENGINE

I. Design - Finished

II. Detailing - Finished

III. Fabrication

Fabrication of the experimental engine is approximately 75 percent complete. The currently estimated completion date is December 23, 1962.

A new test type cylinder head has been designed and released for fabrication. This cylinder head has a port which will accept a standard Photocon balanced pressure pick-up and cooling adapter. This approach was taken because of suitable standard pick-ups could not be found which would fit the existing 10 mm pick-up port, and, because of the expense and uncertainty of fabrication time required to acquire a special pick-up.

IV. Test Facilities

For reasons of economy and scheduling, it has been decided to use the existing Vickers engine test stand (this stand previously used for testing the ASD Hydrox engine). This facility is essentially complete for the purposes of this program except for a new strain gauge torque load cell (of lower range and greater accuracy than the existing one) and a means of taking balanced pressure indicator diaphragm data. It has not yet been decided whether an American Instrument recorder will be used or the point by point procedure employed.

V. Preliminary Tests (Currently Planned)

Should fabrication of the regenerator be completed before the fabrication of the NASA engine, the ASD engine (now available) may be used as an exhaust gas source for preliminary regenerator tests. Should the NASA O_2-Injector be available before the NASA engine, the ASD engine will be modified to use the NASA O_2 Injector.

Tests can be run, in conjunction with analysis, to determine what phase relation between the two hydrogen poppets of the test engine, and what inlet pressure should be used to simulate the flow vs. time characteristics of the single hydrogen poppet flight design engine. Oscilloscope indicator diagrams will be taken.

When both the NASA engine and the O_2 Injector are available the engine will be functionally checked out and the design improvements verified.

VI. Performance Testing Using NASA Engine (Currently Planned)

Steady state performance tests will be run to determine friction, heat rejection, SPC, combustion characteristics, and to obtain balanced pressure indicator diagrams at different power levels.

Power levels will be set by adjusting valving and inlet pressure to simulate the variable phasing engine cycle resulting from the flight engine cycle analysis. Tests with preheated hydrogen will be run using the regenerator. Oscilloscope indicator diagrams will be taken simultaneously with balanced diaphragm indicator diagrams. These diagrams will be accompanied with oscilloscope diagrams obtained using the insulated cylinder head.

If time permits a Quartz cylinder head and mirror will be used to visually evaluate the effects of oxygen injection angle upon mixing and combustion gas swirl. Different oxygen injector jets will be tested to determine the best angle for minimum swirl (to reduce heat rejection) and best mixing.

EXPERIMENTAL REGENERATOR

I. Design and II. Detailing

The analysis, design study, assembly drawing, and detail drawings have been completed for both the extended surface and prime surface configuration of the exhaust gas to hydrogen regenerator. The extended surface configuration was described in Progress Report PR 91565-430-4 for the month of October, 1962.

The working drawing (combined detail and assembly) of the prime surface configuration is shown in Figure 16. The analysis upon which the prime surface design is based is in Appendix A. The prime surface regenerator consists of two concentric, cylindrical stainless steel shells; five (5) concentric coils of 1/4 in. O.D., .016 in. wall, stainless steel hydrogen tubes (filling the annular space between the two concentric shells); an exhaust gas inlet duct with connection flange; an exhaust gas outlet duct; two identical hydrogen tube manifolds; and twenty-one (21) thermocouples.

Each of the hydrogen coils have 40 turns. The concentric spacing of the coils is maintained by 1/16 in. diameter rods spaced 120° apart. The pitch of the coils and the pitch relationship between the coils is maintained by copper strips inserted into the space between the coils at an angle with the axis. The copper strips each have four holes through which the concentricity spacer rods are passed. All connections of the hydrogen tubes are heliarc welded. The outside shell is provided with a window at the cold end for visual inspection of icing.

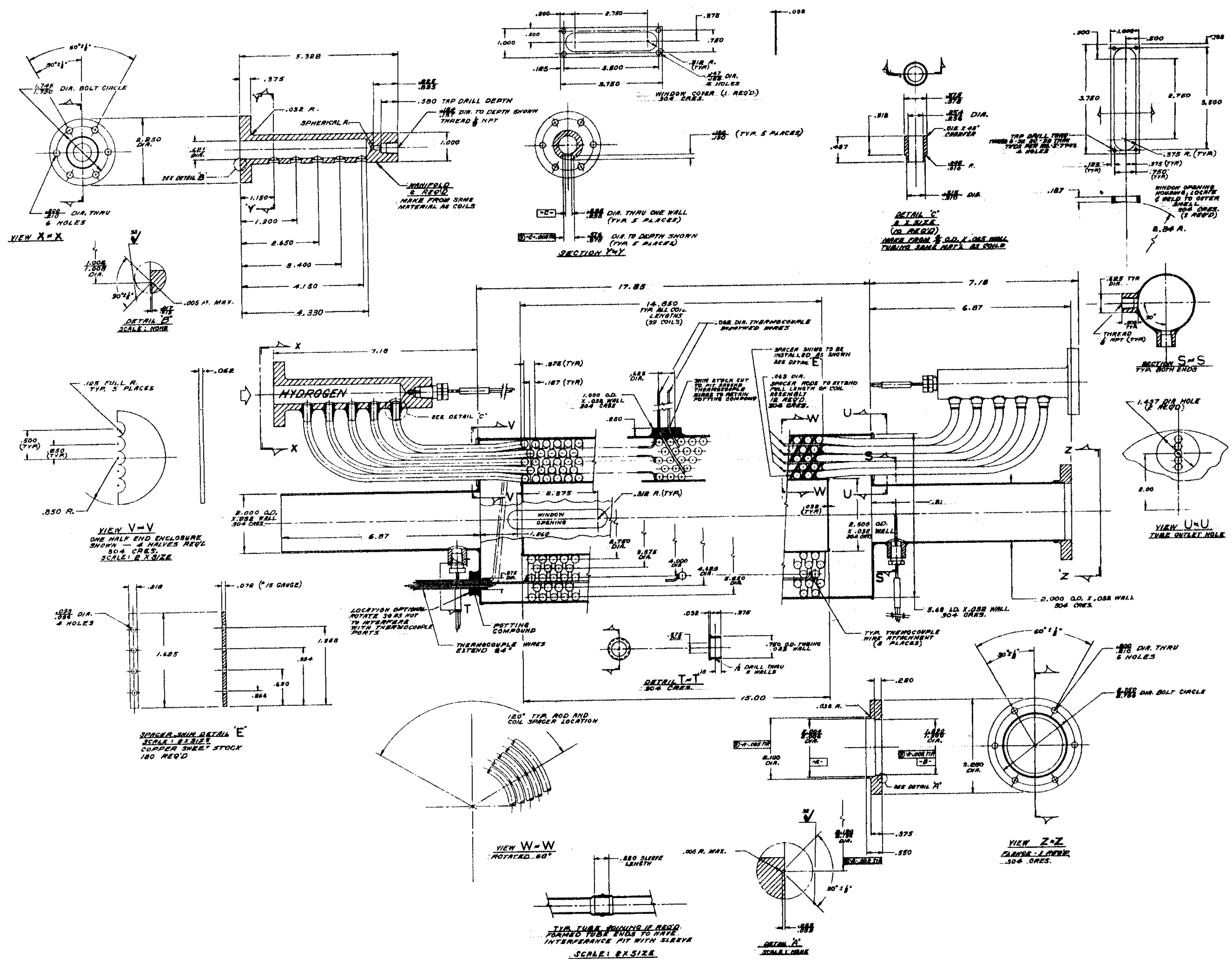


Fig. 16 - Prime Surface Heat Exchanger

Thermocouples are provided for:

H_2 outlet temperature
Exhaust gas inlet temperature
Middle H_2 tube wall temperature at the hot end
Outside H_2 tube wall temperature at the center of the regenerator
Middle H_2 tube wall temperature at the center
Inside H_2 tube wall temperature at the center
Exhaust gas temperature at two locations at the center of the regenerator
Middle H_2 tube wall temperature at the cold end of the regenerator
Exhaust gas outlet temperature
H_2 outlet temperature

Each of the H_2 wall temperature thermocouples will have two back-up (i.e. a total of three (3) couples at each desired temperature location). The regenerator will be packed in suitable insulation during testing.

III. Fabrication and Procurement and IV. Assembly Check-out

Fixed price and fabrication time quotation have been received for both the extended surface and prime surface regenerator configurations. The prime surface configuration was selected for fabrication because it was considerably less expensive, required less fabrication time and is more compact than the extended surface configuration.

A purchase order was given to Sanderson Aircraft Co., 1210 S. Prairie Avenue, Hawthorne, California for one (1) prime surface regenerator per Vickers' design on November 28, 1962.

Sanderson Aircraft has promised delivery three weeks after receipt of order.

Vickers will supply all thermocouples. The H_2 tube wall thermocouples which will have pre-welded junctions and which will be brazed to the H_2 tubes will be inspected by Vickers during assembly.

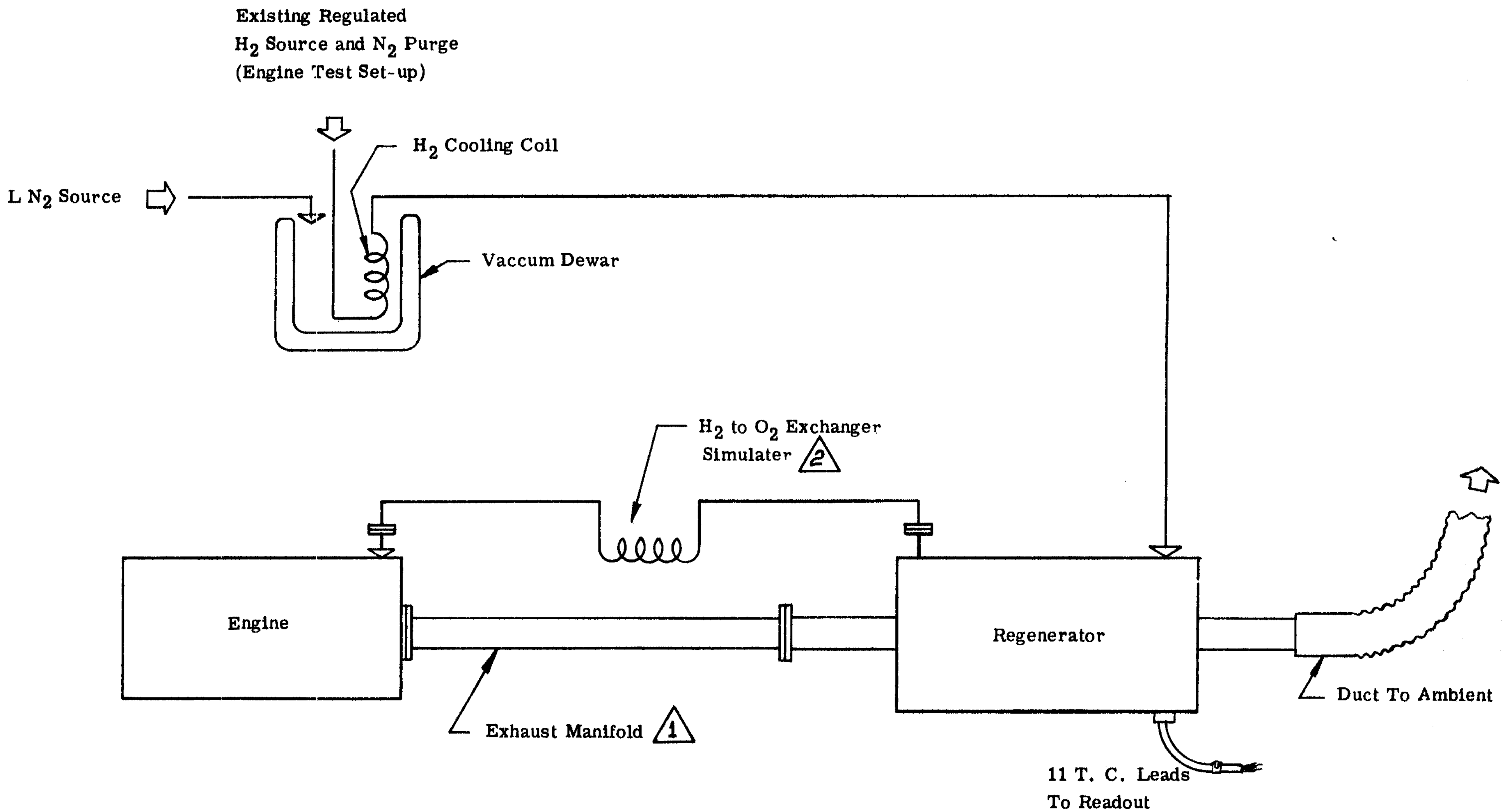
V. Test Facilities

The planned regenerator set-up is shown schematically in Figure 17. The regenerator will be run in closed circuit with engine. The existing facilities of the engine test cell will be used extensively. The additional set-up required consists of the following:

1. Plumbing on insulated line from the liquid nitrogen source to the engine cell.
2. Fabricating an H_2 cooler consisting of a stainless steel coil immersed in a stainless steel purchased vacuum Dewar (to contain the LN_2).
3. Plumbing an insulated stainless steel line with heliarc welded joints from the cooler to the hydrogen inlet port of the regenerator.
4. Constructing an insulated manifold connecting the regenerator to the engine exhaust port.
5. Constructing a manifold connecting the regenerator to the engine hydrogen inlet port. This manifold will be made to simulate the length and volume of the hydrogen side of

Fig. 17

REGENERATOR TEST CIRCUIT SCHEMATIC



the hydrogen to oxygen heat exchanger described in Progress Report PR 91565-430-4 for the month of October, 1962. During initial testing this manifold can be run without insulation so that the engine will not be exposed to the high hydrogen gas temperature prematurely.

6. Insulating the regenerator.

7. Connecting the thermocouple to read out instrumentation.

VI. Preliminary Testing

Proof pressure and leakage testing of the regenerator will be performed by Sanderson Aircraft Co. to Vickers satisfaction during fabrication.

VII. Performance Tests (Currently Planned)

A. Pressure drop tests will be run with ambient nitrogen before the regenerator is installed in the test set-up. Pressure drop vs. flow data will be taken for different hydrogen exit pressures (particularly 300 psia and 1200 psia). This data will be converted to plots of curves of pressure drop vs. flow for hydrogen at various pressures.

B. Heat Transfer Tests - The regenerator test will be run in conjunction with engine test. Various flow rates and pressure levels will be run as compatible with engine testing. The exhaust gas flow rate and composition will be calculated from the oxygen and hydrogen flow rates. The calculation will be checked by analysis of the exhaust gas if time permits. Sufficient temperature data will be taken to determine the hot end and cold end approach temperatures, and to calculate the film coefficients at each end and the middle of the regenerator.

Visual data of ice build-up and for evidence of contamination due to engine lubrication will be taken.

FUTURE PLANS

The parametric studies and control system studies will be completed during the next report period. Testing will begin on the hydrogen compressor and the regenerator.

REFERENCES

1. Taylor C. F., the Internal Combustion Engine in Theory and Practice, Vol. 1, Wiley & Sons, 1960, pp. 171 - 177 and pp. 288 - 290.

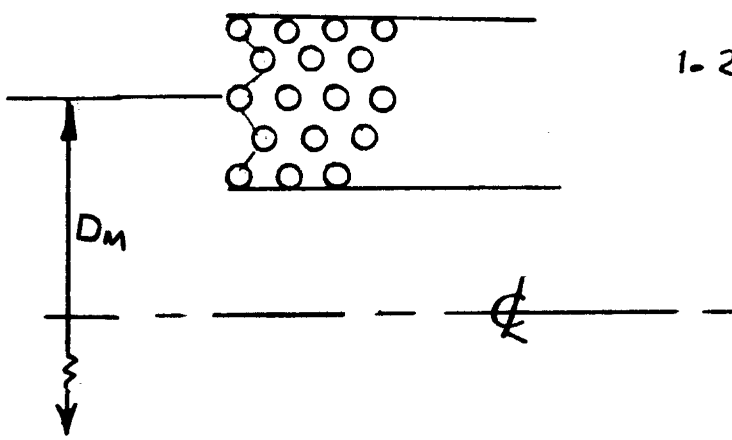
2. Lichty L. C., Internal Combustion Engines, McGraw - Hill, 1951 (Sixth Edition) pp. 480 - 486.

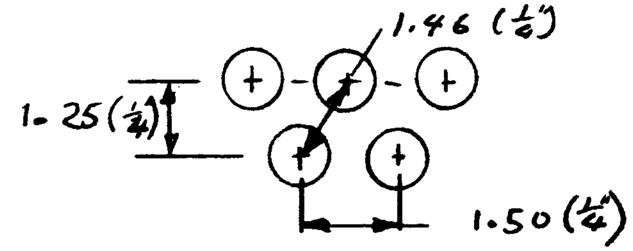
APPENDIX "A"

PREPARED BY A. FINK	VICKERS INCORPORATED ENGINEERING CALCULATION FORM	PAGE 1 OF 5
CHECKED BY & COPIED & RESIZED BY [illegible]	SUBJECT PRIME SURFACE HEAT EXCHANGER	PROGRAM & PROJECT NO. 91565-439
		DATE

ASSUMED CONFIGURATION:

HYDROGEN TUBE 1/4" O.D. - .016" WALL
NUMBER OF COIL SETS - 5
TRANSVERSE PITCH = 1.25(TUBE O.D.)
AXIAL PITCH = 1.50 (TUBE O.D.)
MEAN COIL DIAMETER OF MIDDLE COIL = 4" = .333' = D_M





EXHAUST GAS SIDE:

EXHAUST FLOW AREA = $(4)\,\pi\,(.333)\dfrac{.46}{4\,(12)} = 0.0401\ ft^2$

TRANSFER SURFACE (4 COILS) = $4\left[\dfrac{\pi}{4\times12}\right]\dfrac{ft^2}{ft}\times .333\,\dfrac{\pi\,ft}{coil}\times\dfrac{48\ coil}{1.5\ ft}\times L\ ft$

WHERE L = EXCHANGER LENGTH

$= .833\,\pi^2 L\ ft^2$

$W = 42.3\ \#/HR$ (REQ'D)

$G = \dfrac{42.3}{.0401} = 905.\ \dfrac{\#/HR}{ft^2}$

$$r_h = \frac{A_{flow}}{A_{SURFACE}/L} = \frac{.0401}{.833\pi^2} = .00488 \quad *$$

$$D_h = 4 r_h = .0195'$$

$$N_{Re} = \frac{D_h G}{\mu} = \frac{.0195(905)}{.0286} = 617$$

Let $\frac{h_f}{G C_p} N_{Pr}^{2/3} = .026 \quad **$

$$h_f = \frac{905(1.22)(.026)}{0.79} = 36.4 \frac{B}{HR\,Ft^2\,{}^oF}$$

DESIGNING FOR 58,400 B/HR & $\theta_{fILM} = 150\,{}^oF$

$$A_S = .833\pi^2 L = \frac{58,400}{36.4(150)} = 10.7\ ft^2$$

$$L = \frac{10.7}{.833\pi^2} = 1.3'$$

$$n = \frac{1.3 \times 12}{.375} \approx 40 \text{ turns / coil set}$$

MEAN TUBE LENGTH = $40\pi D_m$ = 42 ft

** fig 92 pg 50 COMPACT HEAT EXCH.

* Pg 3 COMPACT HEAT EXCHANGER BY KAYS & LONDON

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HYDROGEN SIDE

$$\frac{W}{N} = \frac{14.1 \text{ \#/HR. (REQ'D)}}{5 \text{ TUBES}} = 2.82 \frac{\text{\#/HR}}{\text{TUBE}}$$

TUBE I.D = .0182

$$G = \frac{2.82}{\frac{\pi}{4}(.0182)^2} = 10,900$$

$$N_{Re} = \frac{(.0182)(10,900)}{.0306} = 6,500$$

$$\text{SURFACE AREA} = 5 \text{ TUBES} \times 42 \frac{ft}{\text{COIL}} \times (.0182)\pi$$

$$= 12 \; ft^2$$

$$\Theta \cong \frac{58,400}{12 \; h_f}$$

$k_{H_2} = 0.15$ $\qquad N_{PR} = \mu C_p / k$

$\mu_{H_2} = .03$

$D = .0182$

$$h_f = .0225 \frac{k}{D} \left(\frac{DG}{\mu}\right)^{0.8} \left(N_{PR}\right)^{0.8}$$

$$= .0225 \left(\frac{.15}{.0182}\right)\left(6,500\right)^{.8}(.75)$$

$$= 156 \; B/HR. \; ft^2 \; °F$$

$$\Theta_{FILM} = \frac{58,400}{12\,(156)} = 31 \; °R$$

$$\text{Log}_e \text{ MEAN } \Delta T \cong 150 + 31 \simeq \underline{181 \; °R} + \text{WALL } \Delta T$$

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PRESSURE DROP

EXHAUST GAS SIDE

$$\Delta P \cong \left(\frac{G}{3600}\right)^2 \frac{f L F_s}{2g F_2 \rho_m}$$

{EQ 15 Pg 7 ANALYTICAL METHOD FOR COMPACT REGEN, REPORT No. HWR-130 BY H. WOOD}

WHERE:

$$F_s = \frac{A_{surface}}{Vol\ Exchanger} = \frac{A_{surface}}{L \times A_{FRONTAL\ TOT.}}$$

$$F_2 = \frac{A_{FLOW}}{A_{FRONTAL\ TOT.}}$$

$$\therefore \frac{L F_s}{F_2} = \frac{A_{SURFACE}}{A_{FLOW}}$$

$$\therefore \Delta P \cong \left(\frac{G}{3600}\right)^2 \cdot \frac{f}{2g\rho_m} \frac{A_{surface}}{A_{flow}}$$

FOR $N_{Re} = 617$, $f = .125$ {Pg 50 COMPACT HEAT EXCH.}

$$\rho_m = \frac{P}{RT_m} = \frac{144 \times 2\ PSIA}{257 \times 1000°R} = .0011\ LB/FT^3$$

$$\Delta P = \left(\frac{905}{3600}\right)^2 \times \frac{.125 \times 10.7}{2(32.2)(.0011)(.041)} = 30\ LB/FT^2$$

$$\Delta P = \frac{30}{144} = .208\ PSI$$

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PRESSURE DROP

HYDROGEN SIDE;

$$\Delta P = \frac{G^2(v_2 - v_1)}{\alpha g_c} + \frac{f_m G^2 v_m l}{2 g r_h} \quad *$$

MODIFY TO

$$\Delta P \cong \frac{G^2(v_2 - v_1)}{g_c} + \frac{C_f f_m G^2 v_m l}{2g \; D/4} \quad , (l = \text{tube length})$$

FRICTION FACTOR, $f_m = .009$ FOR $N_{Re} = 6,500$ **

CURVATURE FACTOR $C_F = 2.8$ FOR $N_{Re}\sqrt{\frac{D}{D_H}} = 6500\sqrt{\frac{.0182}{.333}} = 500$ ***

LETING $v_2 = \frac{767 \times 1440}{1200 \times 144} = 6.4$

$v_1 = \frac{767 \times 250}{1200 \times 144} = 1.1$

$\Delta v = 5.3$

ASSUMING $v_m = 7.5/2 = 3.75$

$$\Delta P \cong \left(\frac{10,900}{3600}\right)^2 \frac{(5.3)}{(32.2)} \frac{1}{144} + \frac{2.8 \times .009}{144}\left(\frac{10,900}{3600}\right)^2 \frac{(3.75)\,42}{64.4\left(\frac{.0182}{4}\right)}$$

$$= \underline{\underline{.88}} \text{ PSI}$$

*** fig 6-6 Pg 151 ″

** fig 6-11 Pg 156 ″

* EQ 6-9b Pg 158 McAdams